TR-006-S

TEST REPORT

TRACK-TRAIN DYNAMIC ANALYSIS AND TEST PROGRAM

BARBER S-2 STATIC TEST

February 1976

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FOREWORD

This report, prepared by the Analytical Mechanics Department, Martin Marietta Corporation, Denver Division, under Contract NAS8-29882 presents the results of a test to measure the stiffness and damping parameters of a 70 ton Barber S-2 freight truck. The test was conducted in February, 1976 and was administered by the National Aeronautics and Space Administration, George C. Marshall Space Flight Center, Huntsville, Alabama, under the direction of Mr. Jack Macpherson of the Loads and Dynamics Laboratory. We would like to acknowledge Mr. Ed Lind of the Southern Pacific Transportation Company for providing the test hardware and Mr. Robert Bullock of the Standard Car Truck Company for his assistance during the course of the test.

SUMMARY

This report summarizes the results of a static test of a Barber S-2 freight truck conducted to measure the stiffness and friction parameters of the modes of deformation which are being used in various mathematical models in the railroad industry. The particular truck tested was provided by the Southern Pacific Transportation Company (Mr. Ed Lind) and was in an essentially new condition. Mr. Robert Bullock of the Standard Car Truck Company personally inspected the truck and pronounced it within specification as far as wear of the friction surfaces on the side frames were concerned. He also provided new friction wedges and springs for the wedges.

We experienced some difficulty with the truck hardware since it was in an essentially new condition. The various parts of the truck are castings and have many high spots which cause interference. The normal usage of the trucks wears down the high spots and eliminates the interference. The truck was apparently never used, hence, we had to remove the high spots ourselves. We proceeded with the testing with no difficulty once the interference was removed.

The characteristics of the Barber S-2 are very similar to the ASF ride control truck we had previously tested. This should be no surprise since the construction of the two trucks is very similar. The major difference between the two trucks is the amount of friction between the bolster and side frames in both the vertical and lateral directions. The Barber S-2 has approximately twice the friction the ASF ride control truck has in the fully loaded condition. This does not necessarily imply that all trucks will have this same ratio of friction. It is pointed out that a considerable tolerance on friction could exist, hence the friction data should be used with this in mind. The subject of tolerance on friction is discussed in detail in the report.

A more complete comparison to the ASF truck will be made at a later date. This will include some analysis using the measured data.

INTRODUCTION

The characterization of rail hardware for use in mathematical studies requires testing to measure the parameters which have a dominant effect on the response of the hardware to the operational environment. These parameters take the form of mass, damping and stiffness in either a linear or nonlinear form. Typically the hardware demonstrates a nonlinear character for damping (Coulomb type friction) and stiffness (discontinuous behavior) and the data reduction required puts these nonlinear data in a discrete form for use in the analytical efforts. This report summarizes the testing of a Barber S-2 freight truck conducted to measure these parameters.

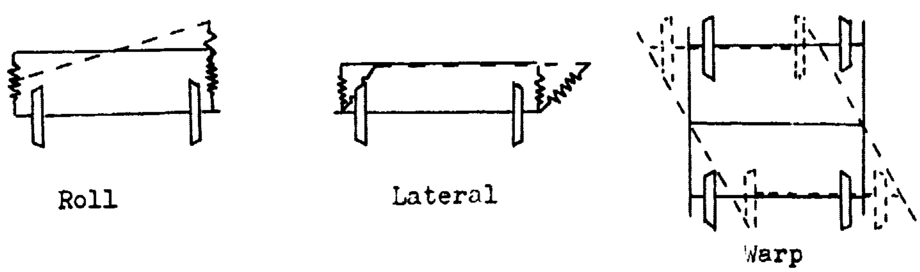
Our previous experience in the test and analysis of the lateral dynamics of an ASF ride control freight truck of the same capacity as the Barber truck (70 ton) indicated that there were a limited number of parameters to measure. The probable modes of distortion of the truck for lateral forces are listed below:

1. roll of bolster with respect to the side frames;

2. lateral translation of the bolster with respect to the side frames;

3. warping of the truck (rotation of the side frames about a vertical axis with respect to the bolster).

The following sketch illustrates these modes of distortion.



Our objectives were, therefore, to measure the damping and stiffness of these modes of distortion so that the rail industry would have the necessary parameters for use in their mathematical models and, hence, would be able to conduct their response studies with a high degree of confidence. Mass data was not obtained in this program as the mass properties for this truck have been well defined in past programs. (Reference 1).

TECHNICAL APPROACH

The physical differences between the ASF and Barber designs are slight; therefore, we took the same basic approach to the measurement of the truck characteristics as we did on the ASF truck test. The exception is that we did not conduct as many tests due to the insight we had gained from the previous test. Previously we conducted six (6) tests. These were:

1. bolster vertical load,
2. bolster lateral load,
3. bolster roll moment,
4. bolster pitch moment,
5. bolster longitudinal load, and
6. warping.

In the present program we retained tests 1, 2 and 6. Test 3 provides redundant information to test 1. Tests 4 and 5 do not provide any meaningful data for the lateral dynamic models.

The characteristics of the truck are dominated by the spring group/friction wedge behavior. The spring groups provide essentially linear load deflection data for vertical and lateral motions of the bolster with respect to the side frames for a large range of deflections. The friction wedge arrangement provides a Coulomb friction type of restoring force which is constant and opposes the relative velocity. This characteristic makes the static testing of the truck very difficult, hence, the tests were conducted by slowly varying the input loads. The load rate was not varied in this test due to the fact that the friction was not a function of load rate for the ASF truck, hence, we assumed it would not be a function of load rate for this truck.

The effect of vertical load was also examined for its effect on the values of the parameters to be measured. The range of vertical loads used covered the range of a completely empty to a completely full car.

The measurements which were made to determine the parameters were loads and displacements. The resulting data acquired was reduced in a form of load versus deflection plots. This data was then interpreted in terms of the nonlinear model which may be used to represent the physical situation as portrayed by this class of trucks.

The remainder of this report will describe, in detail, the test specimen, test setup, instrumentation, data acquisition system and data reduction and interpretation.

Test Specimen

The test specimen was a Barber S-2 which was provided by the Southern

Pacific Transportation Company through the cooperation of Mr. Ed Lind. The external appearance of the truck was very rusty and we became concerned of the actual condition of the truck from a wear standpoint. The casting date for the bolster was 1961 and for the side frames 1968. These dates increased our concerns. Mr. Robert Bullock of Standard Car Truck Company was consulted and he came to our facility to inspect the hardware. Figures 1, 2 and 3 illustrate the results of that inspection. The measurements indicate the hardware was in an essentially new condition. A visual inspection of the surfaces of the friction wedges and the corresponding surfaces on the side frames also indicated that the truck had seen very little use. The friction wedges and their springs were replaced with new hardware provided by Mr. Bullock.

We also conducted some static tests on the springs of the spring groups in order to determine if they were in spec with regard to load and deflection criteria for acceptance or rejection. Table I gives the criteria and the actual measurements for each spring.

We weighed each major component with the following results.

Component	Weight (lbs)
Bolster	1050
Side Frame	850
Side Frame	850
Axle	2400
Axle	2400

Test Setup

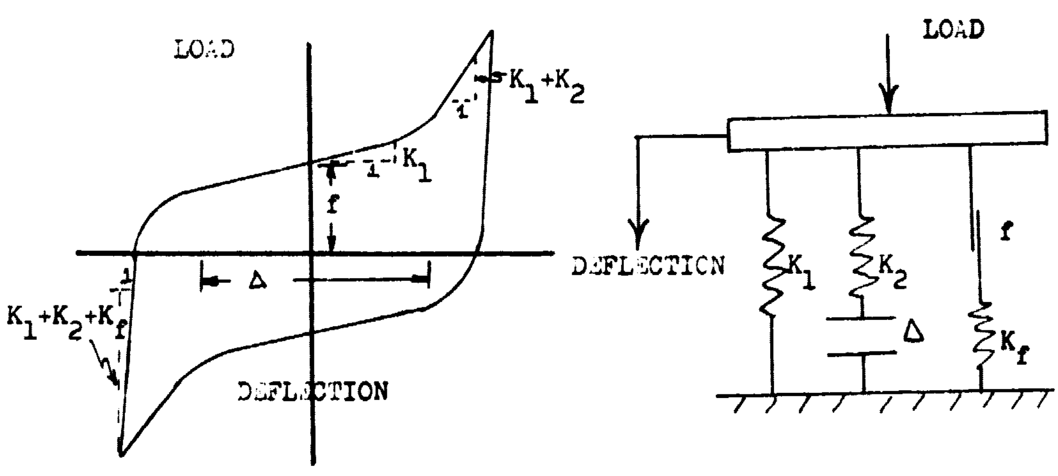
The test setupsfor tests 1, 2 and 3 are illustrated in Figures 4, 5 and 6. The truck was physically inverted and supported on the bolster center plate. The inverted position was used since it allowed conducting all three tests without having to change the truck position. Table II lists the loading conditions for all tests. In the vertical test, the vertical load was the only load applied. The purpose of this test was to measure the roll stiffness and friction of the spring groups. Notice that the vertical test provides the necessary definition of the roll stiffness due to the fact that the local rotational stiffness of the spring groups is very small. The reason for doing the roll test with vertical loads is for ease of fixturing. The data from this test, along with the geometry of the spring groups, will be used to calculate the effective roll stiffness and friction of the bolster with respect to the side frames. In subsequent tests the vertical load was a constant load to simulate the weight of half of a car. Vertical loads of 20K, 50K and 100K pounds were used to cover the range of an empty to full car. The lateral test was simply a lateral load applied to the axles in a cyclic fashion. The cyclic rate was .25 cycles per second. Due to the bolster support the lateral load was reacted at the bolster center plate. The warping test consisted of equal and opposite loads diagonally applied to the truck at the ends of the axles. The applied load is theoretically an equilibrium load set, hence, no reaction would be expected at the bolster center plate. No effort was made to measure this reaction.

The instrumentation consisted of load and displacement transducers. Figure 7 illustrates the locations of the transducers. Table III tabulates the actual locations in the coordinate system shown on Figure 7. All the cyclic load and deflection data was recorded on magnetic tape for later reduction. The instrumentation schematic for the data acquisition system is illustrated in Figure 8.

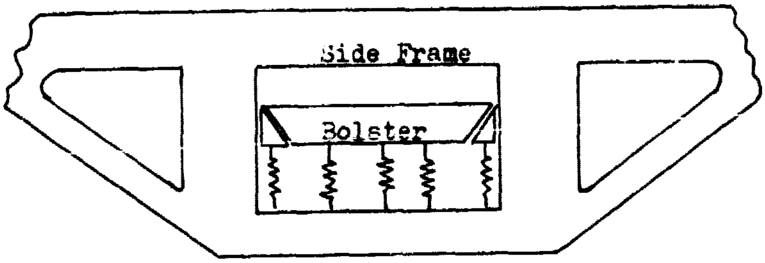
The detailed test procedure and test log is given in Reference 2.

Data Reduction

The data was reduced by simply making cross plots of load versus deflection. The schematic for this reduction technique is illustrated in Figure 8. Having the data in this form it is a relatively simple matter to interpret it in the light of the mathematical parameters for which we are looking. The sketch below illustrates the expected general form for load deflection data in the presence of friction and nonlinear stiffness and the discrete parameter model by which it is represented.



The Barber design does not yield to this form directly in the vertical direction due to the varying friction. The friction wedge is loaded by a spring whose bottom end is connected to the side frame. The sketch below illustrates this configuration.



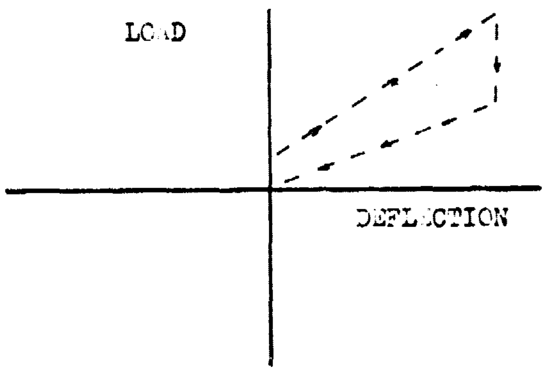
This configuration causes the friction force to increase as the bolster move downward with respect to the side frame. Hence, the equilibrium equation governing this situation is,

$$F_{bolster} = Kx + f(x)\,|V|/V$$

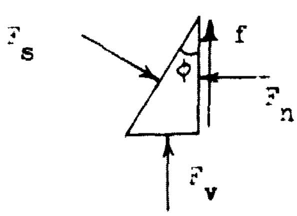
x - relative displacement

V - relative velocity

and the resulting load deflection plot would be expected to appear as shown in the sketch below.

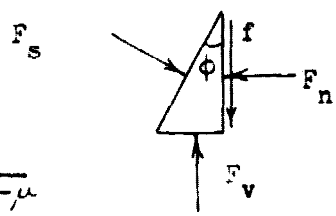


This picture is somewhat deceiving, however, since it appears that the friction force is the same for a motion in either direction. Consider the equilibrium of the friction wedge. Sketch a. below illustrates the equilibrium for a downward relative motion of the bolster relative to the side frame. Sketch b. below illustrates the equilibrium for an upward relative motion of the bolster relative to the side frame. Also shown along side the sketch is the equation for the frictional force between the bolster and side frame. The assumption required for this equation to be valid is the reaction parallel to the sloping surface is zero. This is equivalent to saying there is no friction or coefficient of friction along this surface.



$$f = \frac{\mu F_v}{\mathrm{Tan}\phi - \mu}$$

a.



$$f = \frac{\mu F_v}{\mathrm{Tan}\phi + \mu}$$

b.

The uncertainty involved in the calculation of the friction force is, of course, the coefficient of friction. The value may range between .4 and .8 for steel on steel depending on the condition of the surface. Notice the sensitivity of the frictional force in sketch a. above when the coefficient of friction approaches the tangent of the wedge angle. Hence, when interpreting the test data from a sample of one, the tolerance possibilities must be assessed. The configuration of our truck was such that the coefficient of friction was probably on the high side. The truck had not seen any service and the contacting surfaces were rusty in appearance. We were told by Mr. Robert Bullock of Standard Car and Truck that the appearance of a used trucks friction surfaces would be like that of a mirror.

Figure 9 illustrates the measured load deflection data for the vertical test. The interpretation of this data is aimed at the definition of a discrete stiffness and friction value for a given operating condition. For instance, at the fully loaded condition (100K vertical load) the apparent friction is one-half the width of the load deflection at the maximum deflection. Notice that this is only an average value since the ratio of the friction force due to downward motion to that due to upward motion is given by

$$\frac{\text{Tan}\,\phi + \mu}{\text{Tan}\,\phi - \mu} = 6$$

Keeping in mind that we are looking for the effective roll stiffness we see that during rolling motion one side of the bolster is moving downward while the other side is moving upward relative to the side frame. This results in a natural vertical/roll coupling which is not accounted for in the lateral mathematical models. We will examine this effect in future analyses.

The other parameter of interest is the stiffness. This is the slope of the load deflection curve after removing the slope due to the variation in the frictional force. Consider the following equation of equilibrium.

$$F_{\text{bolster}} = Kx + F(x)\,|V|/V$$

The friction force variation must be calculated based on knowing the stiffness of the spring under the wedge and the coefficient of friction. The coefficient of friction is determined from the maximum friction force in the load deflection curve. Based on these conditions the effective vertical stiffness of the spring groups is 45,000 pounds per inch. This correlates very well with the sum of the individual spring stiffnesses which were measured to insure the springs were withir specified limits for load and deflection.

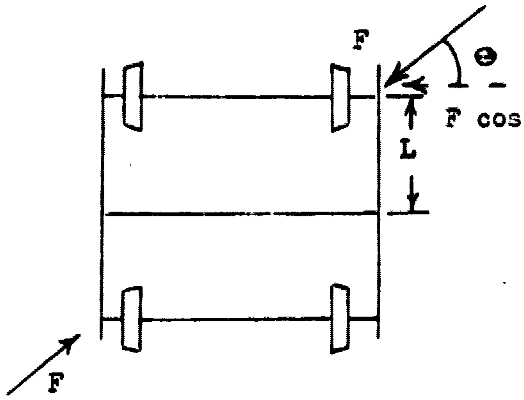
The conversion of the stiffness and friction measured for the vertical direction to the effective roll stiffness and friction simply requires the knowledge of the distance to the center of rotation of the spring groups. The following equations yield these effective values.

$$K_{roll} = K_v (L/2)^2 \qquad \text{where } L = \text{distance between side frames } 78''$$

$$f_{roll} = f_v \, L/2$$

The interpretation of the lateral stiffness and friction data is much more straightforward than for the vertical case. In this case the vertical deflection is constant and the lateral friction force is constant. Figure 10 illustrates the load deflection data for the lateral test. Again the friction is one-half the width of the curve and the stiffness is the slope of the curve as shown on the curve. For the particular case of the Barber truck we had to modify the hardware in order to be able to carry out the test. The maximum force available for the lateral direction was 10K pounds. The friction force was so high that we were unable to cause the truck to deform in this mode with the available force. Hence, in order to measure the spring group stiffness in the lateral direction required the lubrication of the friction surfaces. Therefore, the load deflection data may not be used directly. The apparent friction must be calculated from the coefficient of friction data obtained from the vertical test. The absolute values of the friction as measured or calculated are not particularly important in themselves since the possible range of values is large. The calculated frictional values for the lateral degree of freedom for various load conditions are given in Table IV. This table also gives the stiffness and friction data for the other degrees of freedom (roll and warp).

The warping test data like the lateral is relatively easy to interpret. The friction wedges were not lubricated for this test as the available force was adequate to overcome the friction for this mode of deformation. The interpreted values of stiffness and friction for the warping mode are obtained by the following consideration of the generalized force in the warping mode due to the applied forces. This interpretation requires the assumption that the mode is a pure rotation of the side frames relative to the bolster about a vertical axis. The generalized force is then the moment of the applied force about the bolster/side frame contact point or point of rotation. The dominant force is the component of the applied force along the axle times the distance to the point of rotation. This is illustrated in the sketch below.



The warping degree of freedom is basically a rotational degree of freedom which we measure with linear deflections. The geometry of the instrument used to measure this degree of freedom also enters into the calculation of the warping stiffness and friction. The following equations illustrate this calculation.

angle $\theta = \delta / L$

moment $M_w = 2\, F \cos\theta\, L$ Breakaway moment

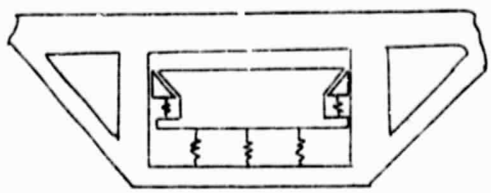
stiffness $K_w = M_w / \theta = 2 \frac{F}{\delta} \cos\theta L^2$

Table IV gives the interpreted values for the warping stiffness and friction. Figure 11 illustrates the warping data.

One subtle aspect of the interpretation of the load deflection data has to do with the slope of the load deflection data as the relative motion reverses sign. Consider the friction phenomenon as it exists between two rigid surfaces. As the relative motion (slipping) proceeds in one direction, a frictional force opposing the motion exists. If an external force is applied to reverse the relative motion, eventually the relative motion (slipping) ceases until the external force is large enough to cause the motion to reverse. During this period of time when the relative motion is zero or there is no slipping, the relative displacement between the two bodies is zero. For the case when there are two elastic bodies in contact there can be relative displacement between the two bodies when there is no slipping. Therefore, the slope of this part of the load deflection curve is also measured and tabulated in Table IV. This parameter may not turn out to be a significant one but is tabulated for later study.

<u>Comparison to ASF Truck</u>

As stated earlier, the physical differences between the ASF and Barber S-2 trucks are slight. The major difference is the configuration of the friction wedge. In the ASF design the spring that preloads the wedge is contained in the bolster as shown in the sketch below. In this case the friction force is independent of the vertical displacement or load on the bolster. The friction force in the Barber design is proportional to the vertical displacement. Other than that configurational difference the two designs are very similar and physically they behave very similarly. Table IV summarizes the stiffness and friction values for the two trucks.



It should be emphasized once again that the tolerance on the truck parameters, especially the friction should be taken into account when doing any analysis with the data. In particular, the analytical effort should establish the sensitivity to the high tolerance parameters and, hence, draw the appropriate conclusions accounting for that sensitivity.

The friction values for all degrees of freedom should be treated as high tolerance parameters as should the warping stiffness. The vertical and lateral stiffnesses are relatively insensitive to the condition of the truck.

Conclusions

This report has summarized the results of a test to measure the stiffness and friction parameters of a Barber S-2 70 ton freight truck for a limited number of modes of deformation. These modes of deformation are:

1. bolster roll;
2. bolster lateral translation relative to the side frames;
3. warping-side frame rotation relative to the bolster about a vertical axis.

The stiffness characteristics of modes 1 and 2 are controlled by the stiffness of the spring groups and should not show a large variation from truck to truck. The physics of the stiffness of the warping mode is not well understood by this author but is thought to also be controlled by the spring groups and the action of the friction wedge as the side frame attempts to rotate it about a vertical axis with respect to the bolster. This particular stiffness parameter is, therefore, somewhat sensitive to the condition of the truck components as they wear. Probably this stiffness will decrease as the components wear.

The friction characteristics of the modes of deformation are also subject to change due to the wear of the components. Since our test was conducted on a piece of "new" hardware, the friction values are probably an upper limit for that design. It is suggested that the use of these friction parameters in mathematical studies be tempered with this judgement and that a consideration given to a possible tolerance. This tolerance should be at least a factor of 2. reduction in friction for all modes.

This testing has assumed that there is no interaction between the modes of deformation of the truck, i.e., the stiffness of friction in one mode is not affected by the motion of another mode. This is probably not a bad assumption for stiffness, however, friction is another story. The friction in the roll and lateral modes is completely controlled by the friction wedges. Once the wedges are sliding on the side frame due to the motion of one of the degrees of freedom the effort to slide in another degree of freedom is reduced to a very small amount. Hence, the apparent friction in the real environment may be much smaller than measured in our "uncoupled" test. Testing to excite multiple modes may provide the answer to this question, however, the test setup involved may be considerably more expensive.

It should be emphasized that it is important to have a good understanding of the physics of the hardware that is being tested by the people who are doing the testing. Without this understanding, it is very nearly impossible to conduct a proper test and record the proper data. It would only be a stroke of luck if the proper tests were conducted without at least a simple mathematical model of the hardware. Our previous experience with the ASF ride control truck provided this understanding for the very similar Barber S-2 design. Future testing of rail hardware should combine this mathematical modeling either in the form of a consulting service provided by the contracting organization or an independent modeling effort conducted by the test organization. This also provides for the acquisition and reduction of the data in a logical and useful format and will result in a much more meaningful test.

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REFERENCES

1. Part VII of ASF participation in the Track-Train Dynamics Program dated 1-4-74 "Mass Moment of Inertia of side frame (yaw and pitch) and truck bolster (yaw and rock)"
2. TR-148s-76 Test Procedure- Track Train Dynamics Analysis and Test Program Prepared by A. G. Nemes of the Martin Marietta Corporation, Denver division.

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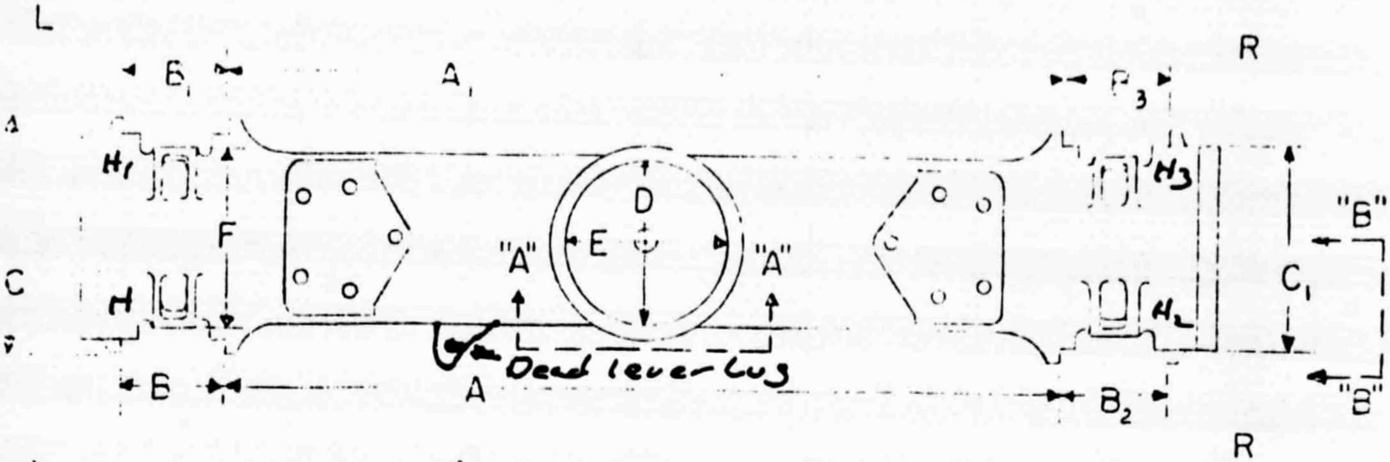
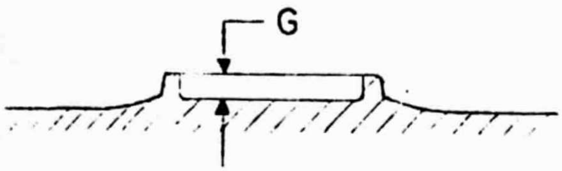


National 37831-17

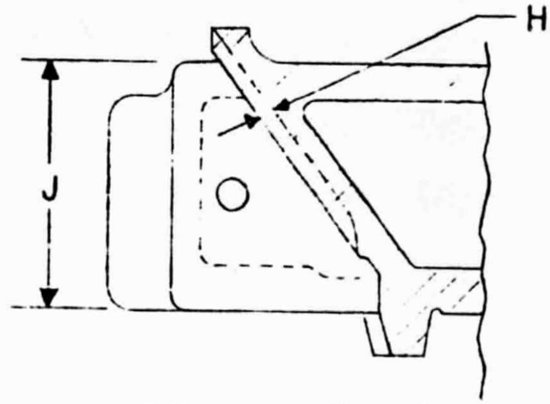
12-61 4870

1 SPF 1600

1050 Pounds



SECTION "A-A"



SECTION "B-B"

$k = 11\frac{21}{32} - \frac{3}{8} =$

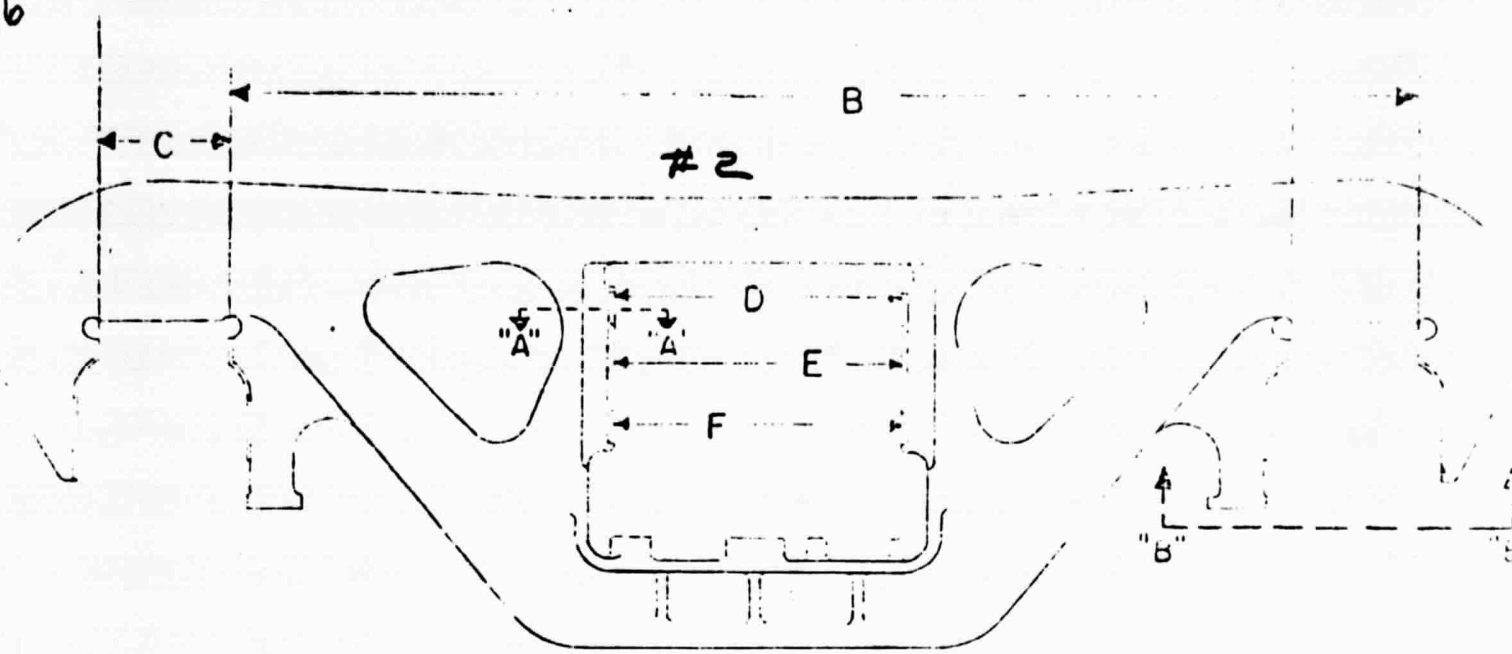
$k_1 = 11\frac{21}{32} - \frac{3}{8}$

ORIGINAL PAGE IS
OF POOR QUALITY

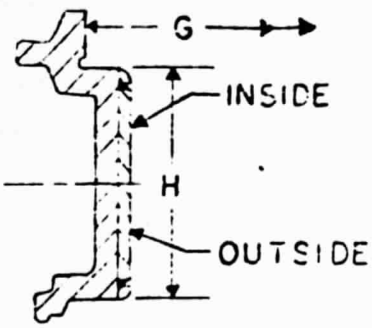
DIMENSION	A	A_1	B	B_1	B_2	B_3	C	C_1	D	E	F	G	H	J
MEASURED	$70\frac{3}{4}$	$70\frac{7}{8}$	$7\frac{7}{16}$	$7\frac{3}{8}$	$7\frac{3}{8}$	$7\frac{3}{8}$	$19\frac{1}{4}$	$19\frac{3}{8}$	$14\frac{1}{16}$	$14\frac{1}{16}$	15	$1\frac{3}{16}$	—	$4\frac{15}{16}$
DESIGN														

H	H_1	H_2	H_3
0"	0"	0"	0"

FIGURE 1. TRUCK BOLSTER INSPECTION SHEET



Gould B TF 7178CP 10-68
F75-04BXI-UA
850 Pounds



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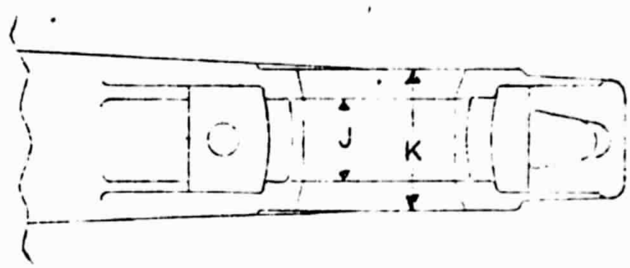
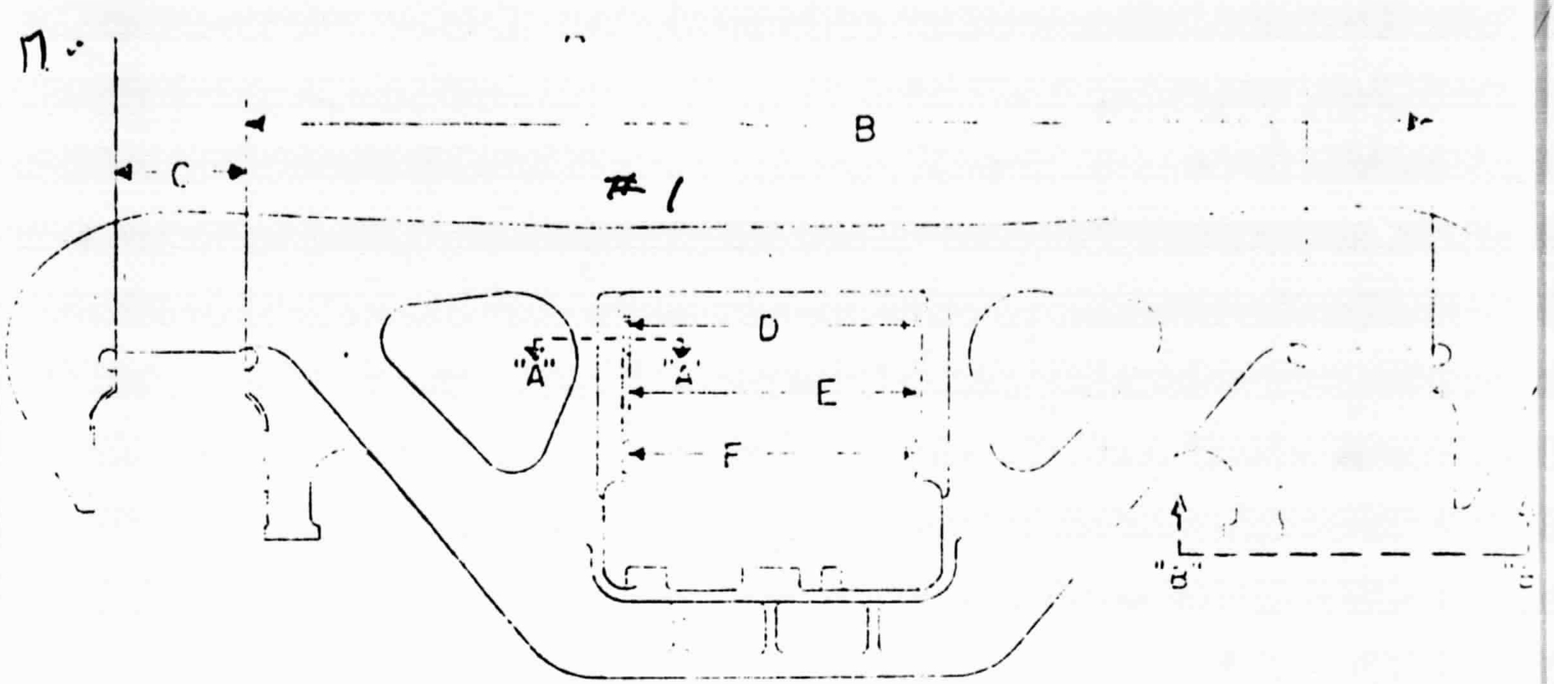


SECTION "B-B"

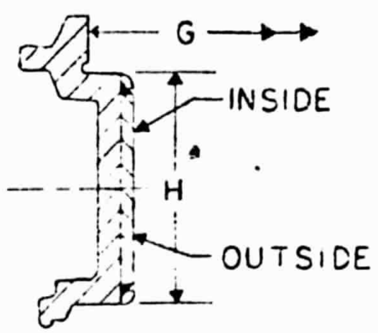
	A	B	C	D	E	F	G	H	J	K
INSIDE	67 15/16	67 7/8	7 3/8	17 1/16	17 1/16	17 1/16	Top 19 3/4	Left 6 1/2	3 3/8	5 7/8
OUTSIDE				17 1/16	17 1/16	17 1/16	Bottom 19 3/4	Right 6 7/16		
DESIGN DIMENSIONS										

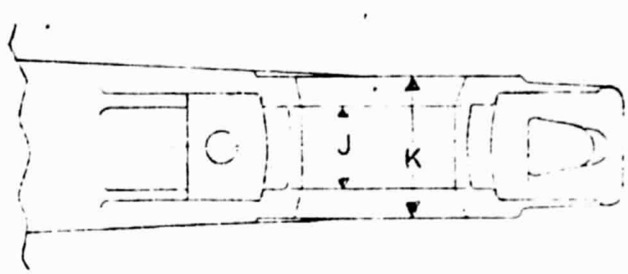
SIDE FRAME INSPECTION
SHEET

FIGURE 2.



Gould B TF 7178CP 10-68
F7S-04BNUA
850 Pounds



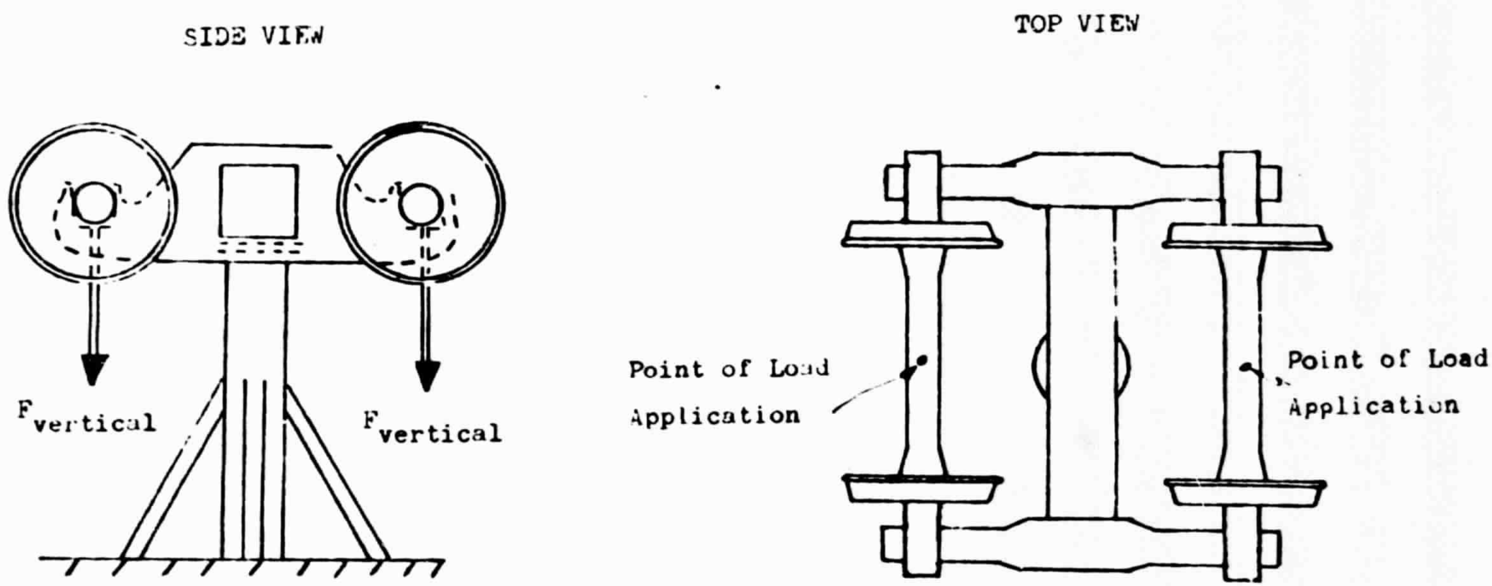


SECTION "A-A"

SECTION "B-B"

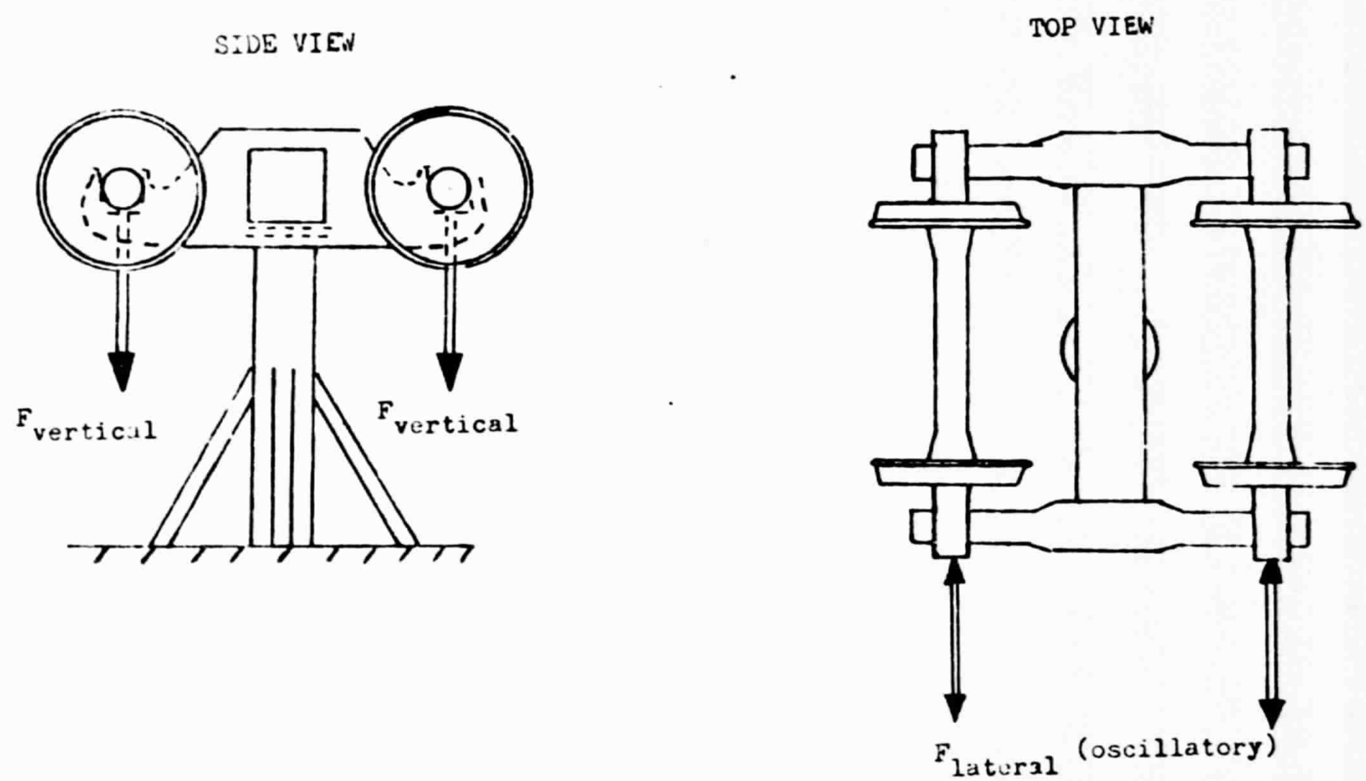
	A	B	C	D	E	F	G	H	J	K
INSIDE	~~68"~~	67 7/8	7 1/4	17 1/16	17 1/16	17 1/16	Top 19 11/16	Left 6 7/16	3 3/8	5 7/8
OUTSIDE	67 7/8			17 1/16	17 1/16	17	Bottom 19 11/16	Right 6 7/16		
DESIGN DIMENSIONS										

FIGURE 3. SIDE FRAME INSPECTION SHEET



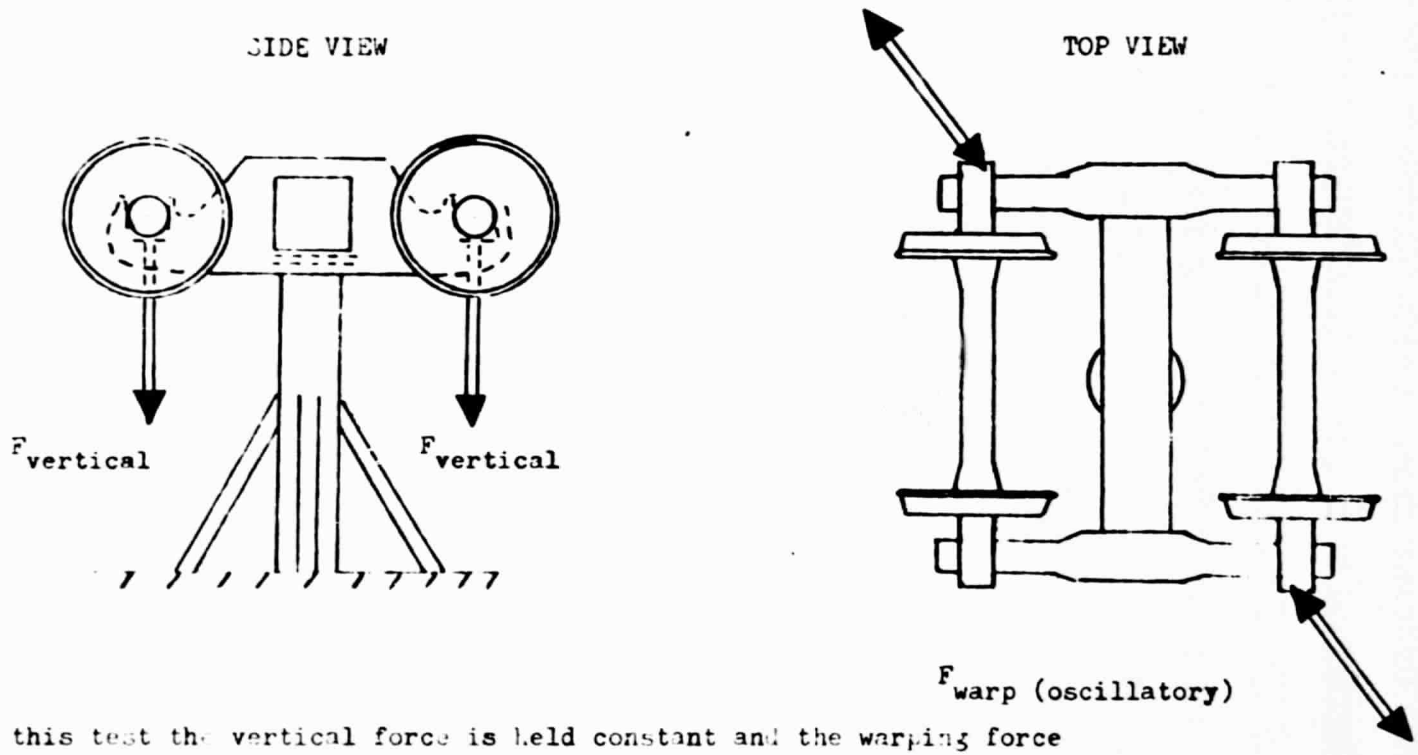
Note: In this test the vertical load was cycled from zero to its maximum value of 100,000 pounds and back to zero.

FIGURE 4. Vertical Test



Note: In this test the vertical force is held constant and the lateral force oscillates plus and minus its maximum value.

FIGURE 5. Lateral Test



Note: In this test the vertical force is held constant and the warping force oscillates plus and minus its maximum value.

FIGURE 6. Warping Test

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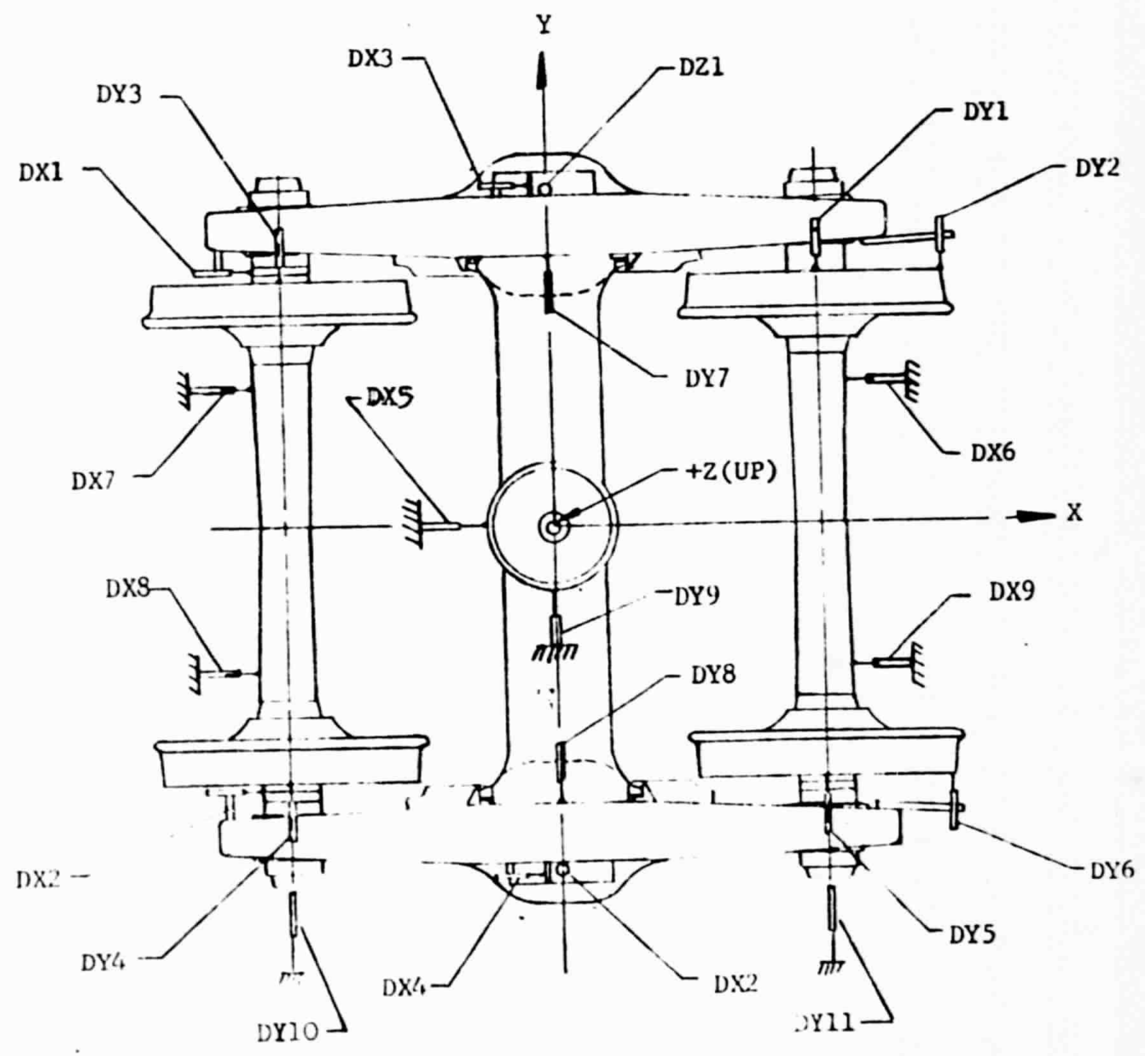


FIGURE 7. Instrumentation Locations

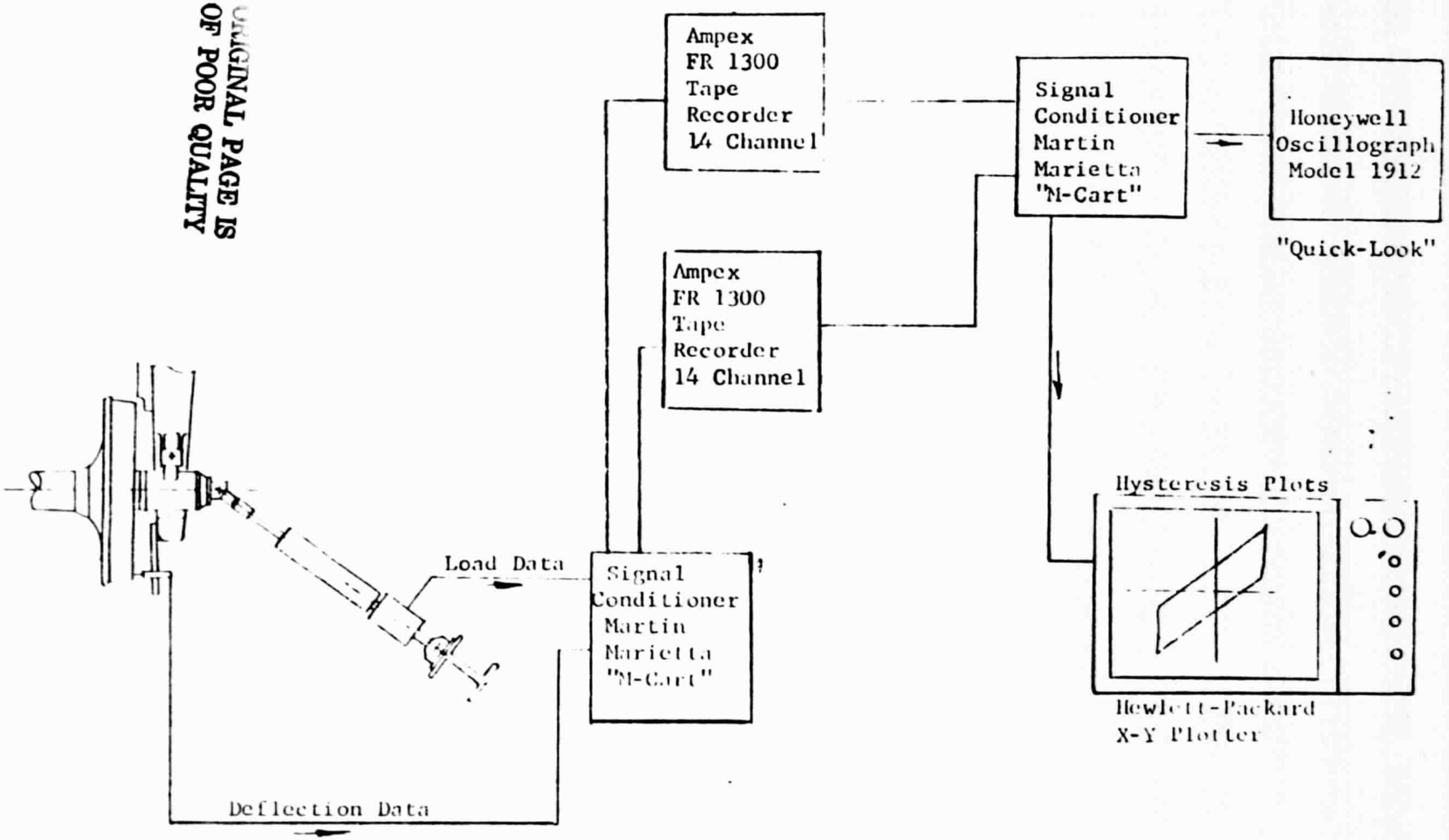


FIGURE 8. Data Acquisition and Reduction Schematic

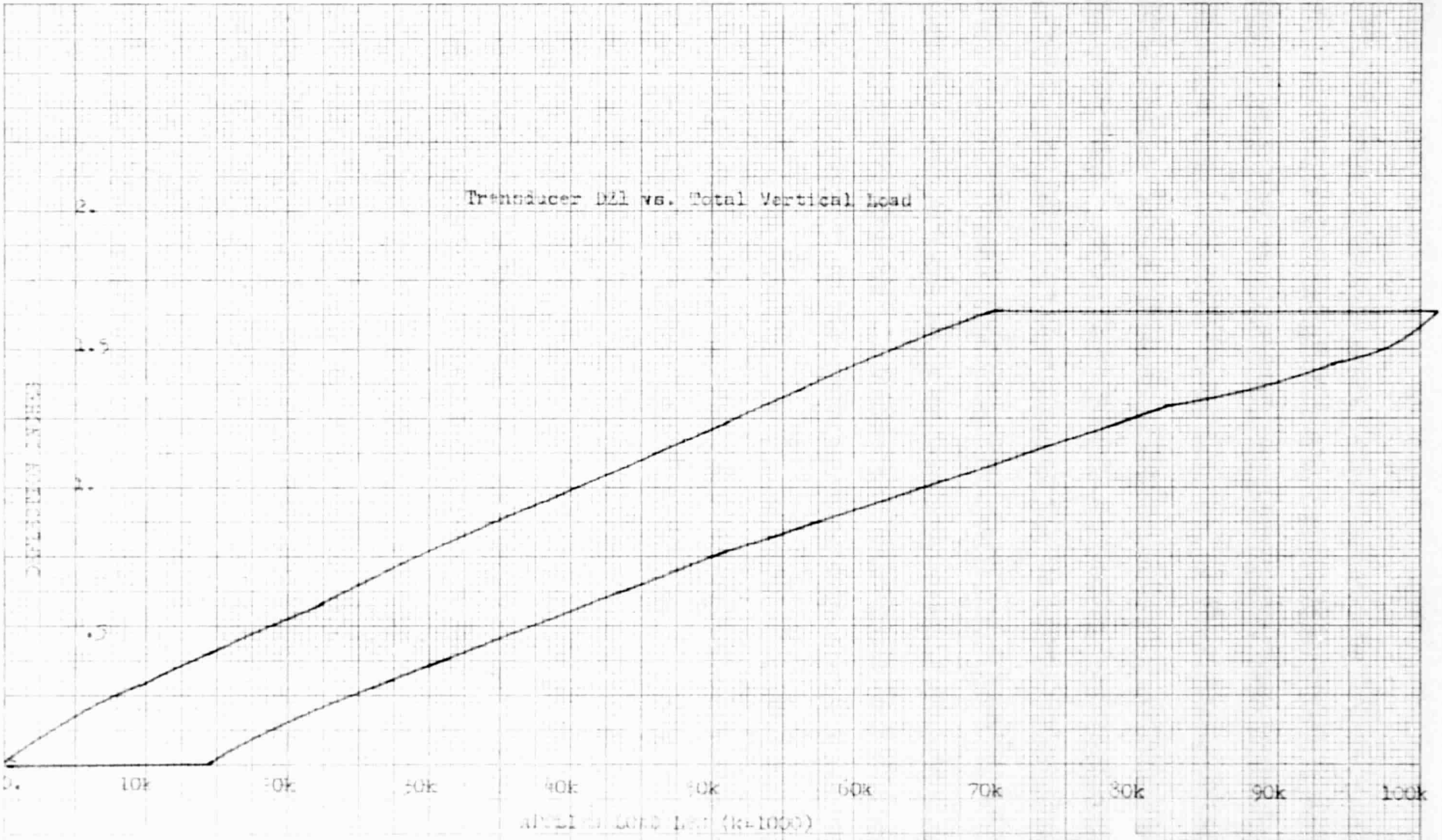


FIGURE 9. LOAD DEFLECTION RESULTS VERTICAL TEST

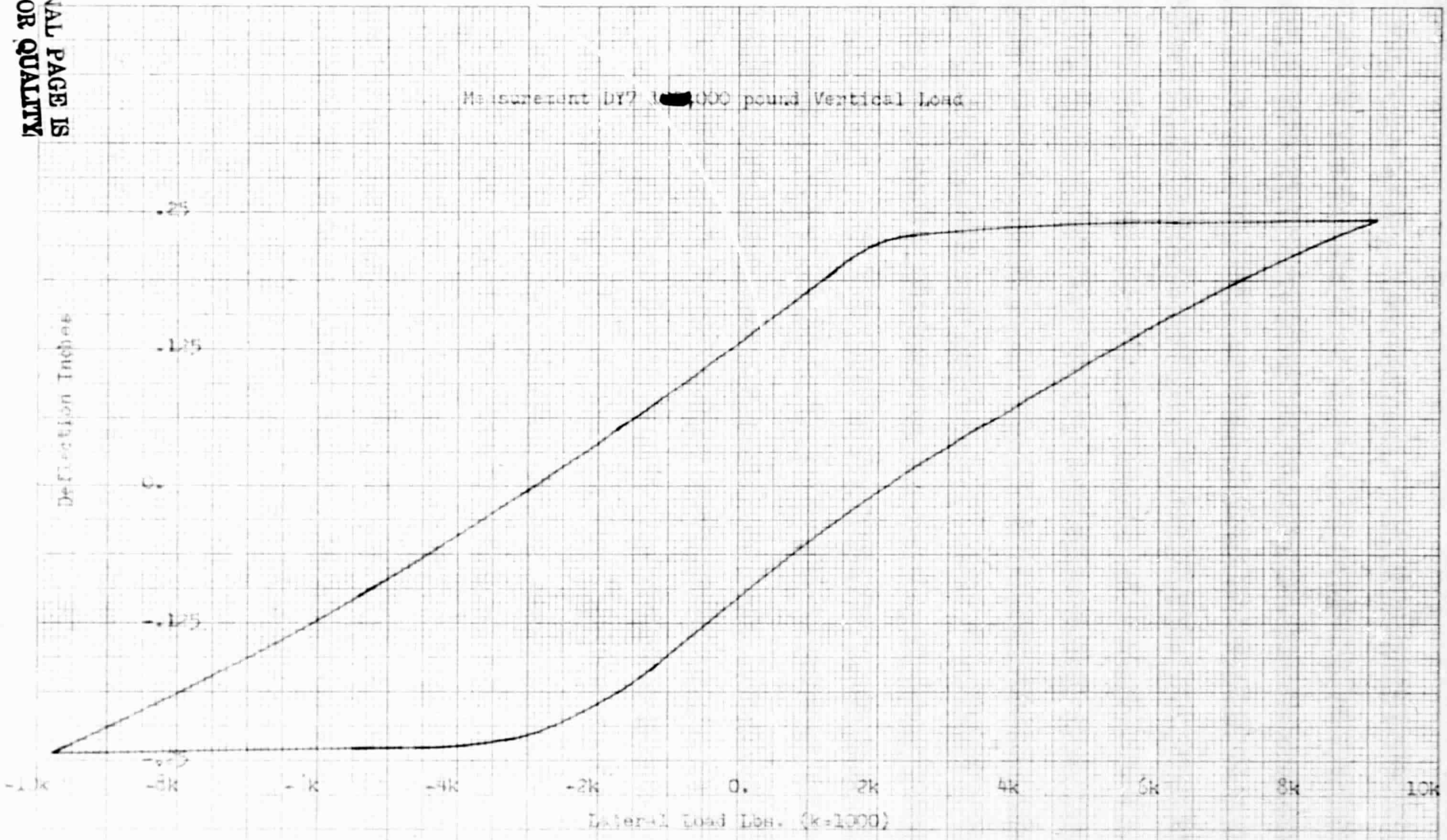


FIGURE 10. Lateral Test Results

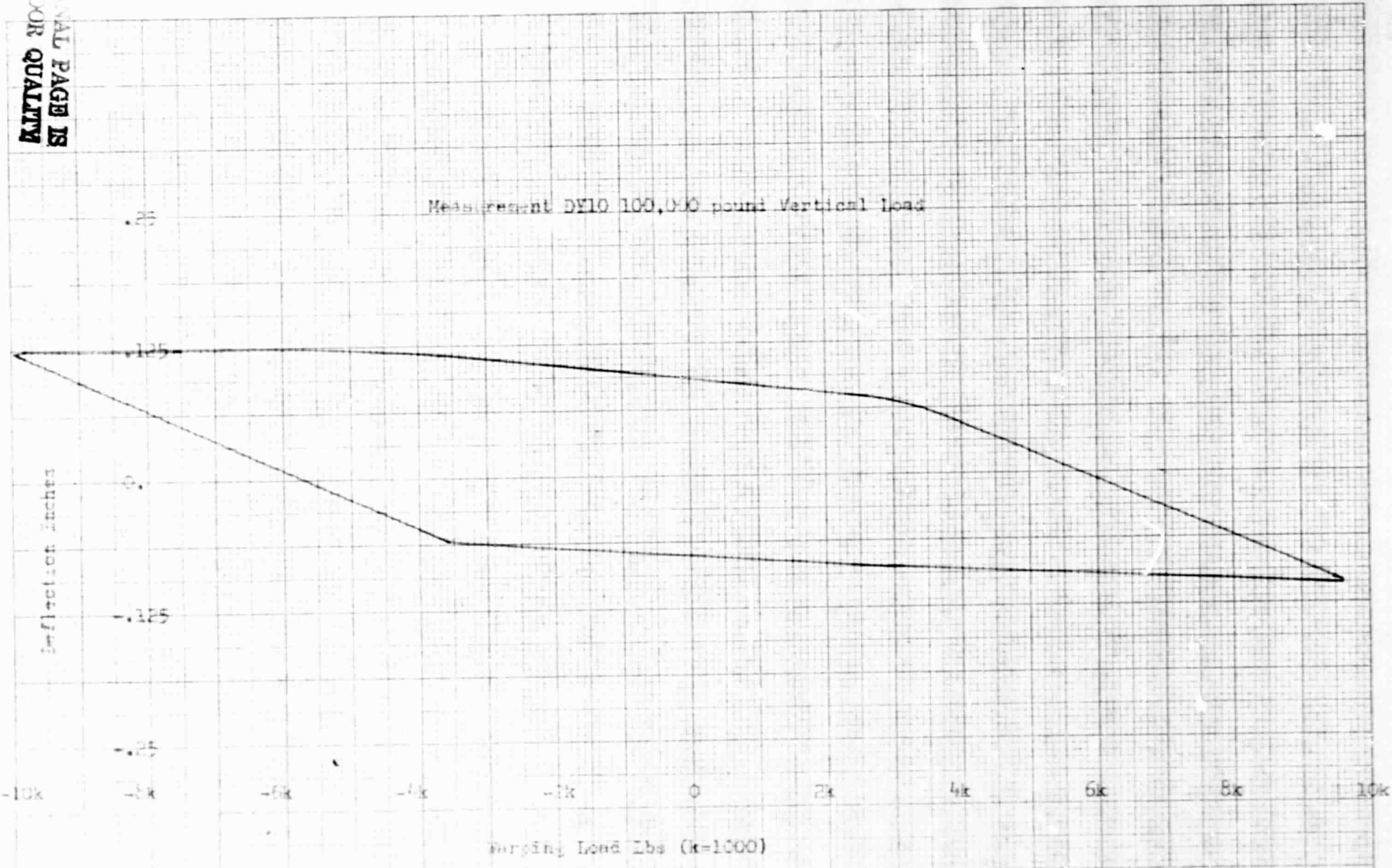


FIGURE 11. Surging Test Results

TABLE I Spring Deflection Data

D-5 Outer Spring Data		
Spring #	Free Height in.	Height @ 5500 Lbs
1	9 31/32	7 21/32
2	10	7 18/32
3	9 30/32	7 1[illegible]/32
4	10 1/32	7 20/32
5	9 23/32	7 6/32
6	9 28/32	7 15/32
7	10 1/32	7 20/32
8	9 27/32	7 15/32
9	9 24/32	7 10/32
10	9 31/32	7 15/32
11	10 2/32	7 24/32
12	9 25/32	7 13/32
13	9 28/32	7 17/32
14	9 28/32	7 16/32

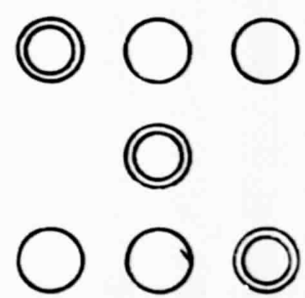
Comdemning Height at no load - 9 20/32

at 5500 lbs- 7 2/32

D-5 Inner Spring Data		
Spring #	Free Height in.	Height @ 2200 Lbs
1	10 1/32	8 2/32
2	10 7/32	8 7/32
3	10 1/32	8 2/32
4	9 29/32	7 30/32
5	10 7/32	8 7/32
6	10 3/32	8 4/32

Condemning height at no load - 9 5/8

at 2200 lbs- 7 21/32



Spring Configuration
6X11 Bearing
220,000 on the rail
7 D5 outers
3 D5 inners

TEST NUMBER	VERTICAL PRELOAD	OSCILLATORY LOAD	LOAD DIRECTION
1	NA	0-100,000-0 lbs	Vertical
2	20,000 lbs	± 2,000 lbs	Lateral
	50,000 lbs	± 5,000 lbs	Through
	100,000 lbs	± 10,000 lbs	axles
3	20,000 lbs	± 2,000 lbs	Diagonally
	50,000 lbs	± 5,000 lbs	Through
	100,000 lbs	± 10,000 lbs	truck

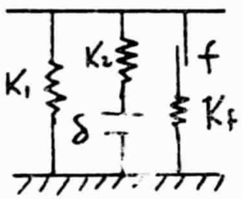
TABLE II LOAD CONDITIONS

Transducer Name	Coordinate Location- inches		
	X	Y	Z
DZ1	0.0	+48 1/2	-
DZ2	0.0	-48 1/2	-
DX1	-	+32 3/4	0.0
DX2	-	-32 3/4	0.0
DX3	-	+33 7/8	*
DX4	-	-33 3/4	**
DX5	-	-1	+9 3/4
DX6	-	+46	0.0
DX7	-	+46	0.0
DX8	-	-46	0.0
DX9	-	-46	0.0
DY1	+34	-	+15 1/2
DY2	+67 3/4	-	- 3/4
DY3	-34	-	+15 1/2
DY4	-34	-	+15 1/2
DY5	+34	-	+15 1/2
DY6	+68	-	**
DY7	0.0	-	**
DY8	0.0	-	**
DY9	0.0	-	+13
DY10	+34	-	+2
DY11	-34	-	+2

* 5 inches above bolster roller pad (+Z) on bolster

** 3 " " " " " " "

TABLE III Instrument Locations



COMPARISON OF BARBER S-2 AND ASF RIDE CONTROL PARAMETER
70 TON CONFIGURATION

PARAMETER

	K_1		K_1+K_2		$K_1+K_2+K_f$ or K_1+K_f		FRICTION (f)		SLOP (δ)	
MODE	BARBER	ASF	BARBER	ASF	BARBER	ASF	BARBER	ASF	BARBER	ASF
VERTICAL	45000#/in	45000#/in	-	-	LARGE >10^6#/in	LARGE >10^6#/in	VARIES 5000# to 15000# OVER RANGE OF VERTICAL LOAD	7000# CONSTANT	SPRING TRAVEL LIMIT	
ROLL	6.84×10^7 in lb/rad	6.84×10^7 in lb/rad			LARGE 10^9 in lb/rad	LARGE				
LATERAL										
VERTICAL PRELOAD 20000 lb	12000#/in	10700#/in	-	-	3.2×10^5 #/in	3.3×10^5 #/in	5000#	6000#	GIB CLEARANCE LIMIT	
50000 lb	16000	15700	-	-	3.2×10^5	3.5×10^5	12000#	6000		
100000 lb	28000	27700	-	-	3.2×10^5	4.2×10^5	15000#	6000		
WARPING	IN#/RAD	IN#/RAD	IN#/RAD	IN#/RAD	IN#/RAD	IN#/RAD	IN#	IN#	DEGREES	
VERTICAL PRELOAD 20000 lb	1.88×10^7	3.43×10^7	-	8.58×10^7	1.6×10^8	2.574×10^8	$.86 \times 10^5$	$.553 \times 10^5$	-	.125
50000 lb	6.02×10^7	3.43×10^7	-	8.58×10^7	3.2×10^8	2.574×10^8	1.66×10^5	1.107×10^5	-	.125
100000 lb	7.53×10^7	3.43×10^7	-	8.58×10^7	4.89×10^8	2.574×10^8	3.18×10^5	1.66×10^5	-	.125

TABLE IV REDUCED DATA FOR BARBER AND ASF TRUCKS